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THE CHARACTERISTICS OF TWO-BLADE PROPELLER FANS

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ABSTRACT

Seven 2-blade propeller fans, 8 feet in diameter and of pitch/diameter ratios 0.250, 0.375, 0.500, 0.625, 0.750, 0.938, and 1.063, respectively, were tested in a duct especially constructed to simulate the operating conditions encountered in cooling towers. Each fan was tested for two operating conditions: (1) With the fan operating as a blower, and (2) with the fan exhausting air from the duct. Each fan was operated at constant speed of rotation against resistance conditions which were varied by steps, from a completely blocked passage to as nearly an open passage as the conditions in the duct permitted. The relation between the volume of air moved in unit time, the head developed, and the power absorbed by the fan was determined for each resistance condition. The results are expressed in the form of coefficients and are plotted so as to facilitate their use in estimating the performance of fans of other diameters and rotational speeds.

The chief effect of variation of pitch/diameter ratio is to increase the range of head and volume coefficients with increase in pitch/diameter ratio up to pitch/diameter ratio of 0.938. The blower condition shows a slightly higher efficiency throughout. There is little variation in maximum efficiency between pitch/diameter ratios of 0.375 and 0.938. Pitch/diameter ratios greater than 1.0 should not, in general, be used.

Two-blade propeller fans are suitable for moving large volumes of air against pressures that do not exceed 1.0 inch of water. For pressures greater than 1.0 inch of water the rotational speeds become inconveniently large. The speed of rotation and the fan diameter should be carefully chosen to obtain maximum efficiency.

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I. INTRODUCTION

The investigation described in this report was undertaken as a cooperative project between the Paragon Engineers (Inc.), of Baltimore, Md., and the National Bureau of Standards for the purpose of obtaining for the industry data on the performance of 2-blade propeller fans. Recently there has been a trend toward the use of fans of this type in cooling-tower installations, and their use has also been considered for other applications where it is desired to move large volumes of air against relatively low pressures. Since no data were available as to the operating characteristics of 2-blade propeller fans, the importance of this information to the manufacturer and user is obvious.

The Paragon Engineers (Inc.) furnished the fans used and supplied funds for the construction of a test duct at the Bureau of Standards, where the experiments were performed. Members of the staff of the Bureau of Standards installed the apparatus and conducted the tests.

This report contains a brief description of the apparatus and methods employed and presents a summary of the results of the investigation, together with such comments and discussion as seem necessary.

II. GENERAL PLAN OF THE INVESTIGATION

In regard to any one fan the object of the experiments is to determine the relation between the power absorbed and the quantity of air delivered against various resistance conditions. When this relation has been found and expressed in suitable form, it can be used to predict the performance of similar fans operating under similar conditions. It is important to remember the restriction stated in the last clause, since the test duct was arranged to simulate the special conditions encountered in cooling-tower applications.

Each fan was operated at constant speed of rotation against resistance conditions which were varied by steps, from a completely blocked passage to as nearly an open passage as conditions in the test duct permitted. The temperature and barometric pressure of the free atmosphere, the static pressure and velocity of the air flowing through the duct, and the power furnished by the motor were measured. The quantity of air delivered by the fan in unit time was calculated from the observed value of the mean air velocity and the area of the duct.

Observations were made on each fan at two speeds of rotation, viz, 585 and 700 r. p. m., except for two fans of low pitch where it was necessary to use speeds of 800 and 900 r. p. m. in order to stay within the efficient working range of the motor. These speeds were selected as representative motor speeds. As will be explained later, the measurements at two speeds are, in effect, check measurements and serve to indicate the accuracy of the work.

Data were obtained for two operating conditions: (1) With the fan operating as a blower, and (2) with the fan exhausting air from the duct.

III. DESCRIPTION OF APPARATUS

The test duct (figs. 1, 2, 10, 11) is circular in cross section, 10 feet in diameter, and 26 feet 3½ inches long, with the axis of the duct about 8 feet above the ground. The duct is out of doors, a control room built at the side of the duct serving to house the instruments and control apparatus. In the duct there is a honeycomb partition consisting of hollow sheet-metal cylinders 3 inches in diameter and 12 inches long with their axes parallel to the axis of the duct. In addition to serving its usual function of straightening the air stream, the honeycomb serves as a support for holding the materials used to increase the resistance of the duct. When the fan is operated as a blower, the honeycomb is located 13 feet 11½ inches from the plane of the fan. (Fig. 10.) For the exhaust condition (fig. 11) the honeycomb is located 21 feet 6⅝ inches from the plane of the fan. Eight static wall plates, 45° apart in a plane normal to the axis of the duct and 13 feet 9⅛ inches from the plane of the fan, are located in the duct wall. The

FIGURE 1.—*Front view of test duct*

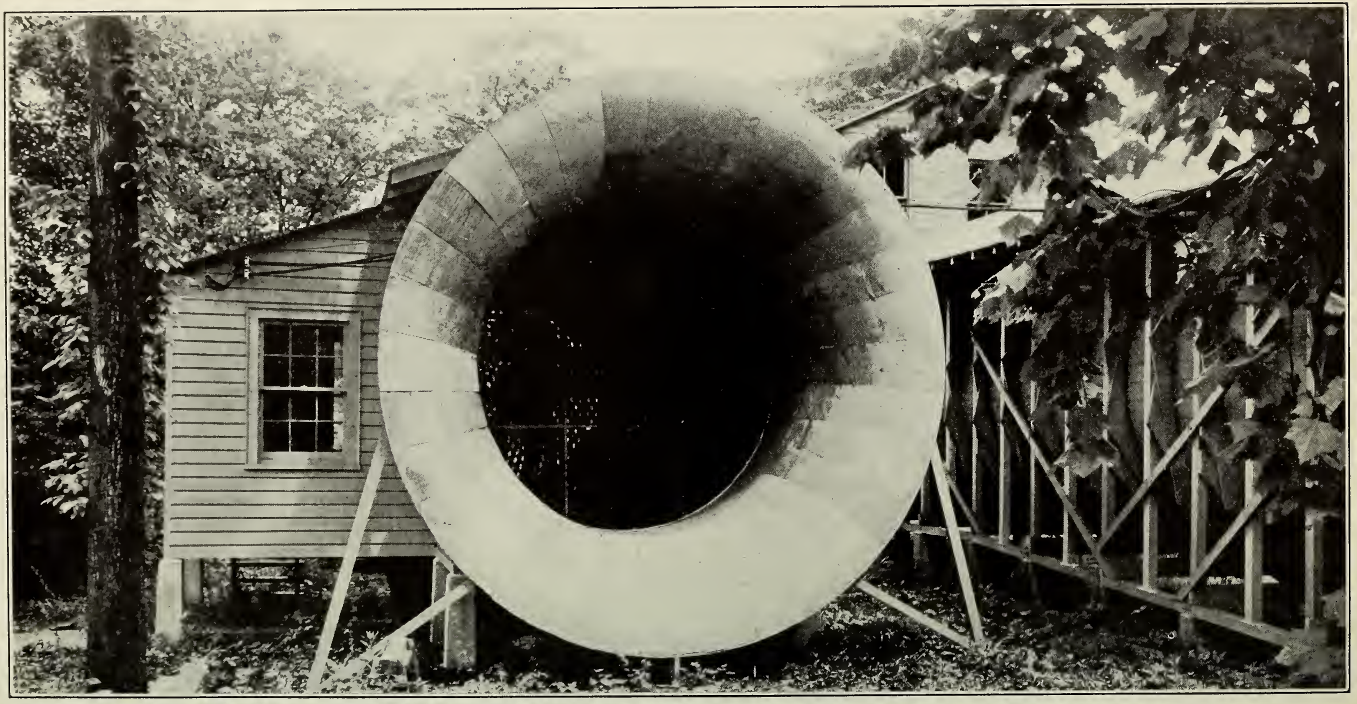

FIGURE 2.—*Rear view of test duct showing honeycomb, faired section, and control room*

wall plates are connected through one-eighth inch galvanized pipe to a single header. Each lead contains a valve so that any wall plate may be connected through the header to a sloping manometer while the rest are closed off.

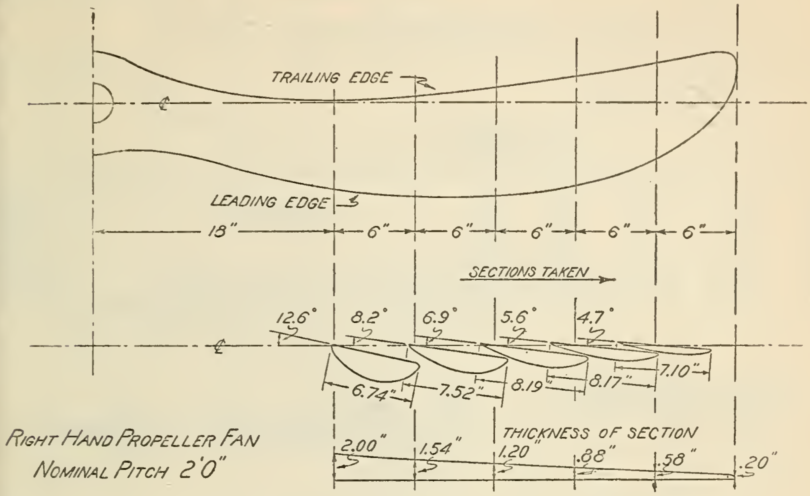

FIGURE 3

A 25-horsepower direct-current variable-speed motor is mounted on two streamlined concrete columns inside the duct. The motor is substantially cylindrical and about 35 inches in diameter. The

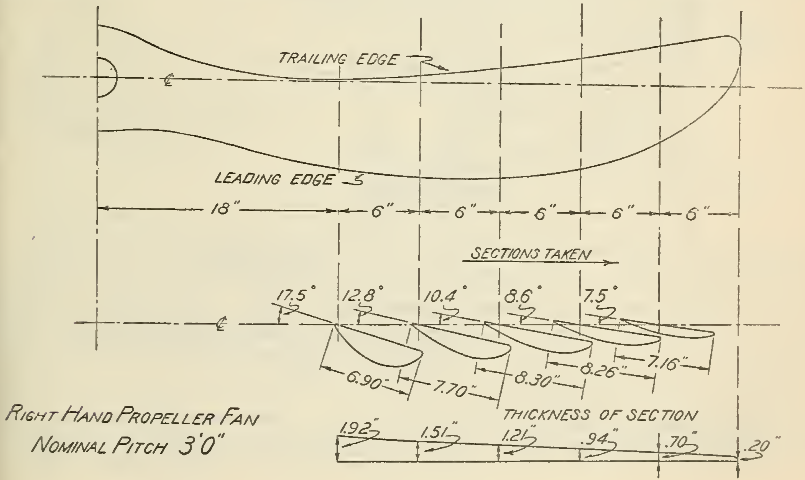

FIGURE 4

streamlined columns are about 3½ feet high and 2½ feet long, with a maximum thickness of about 8 inches.

The air speed at any point was measured by a 4-inch Richard electrically signaling anemometer, which had been calibrated on the mounting used. The anemometer was fastened, as indicated in

Figures 10 and 11, to an arm that could be turned in a plane normal to the axis of the duct, and it could be set at any distance from the axis;

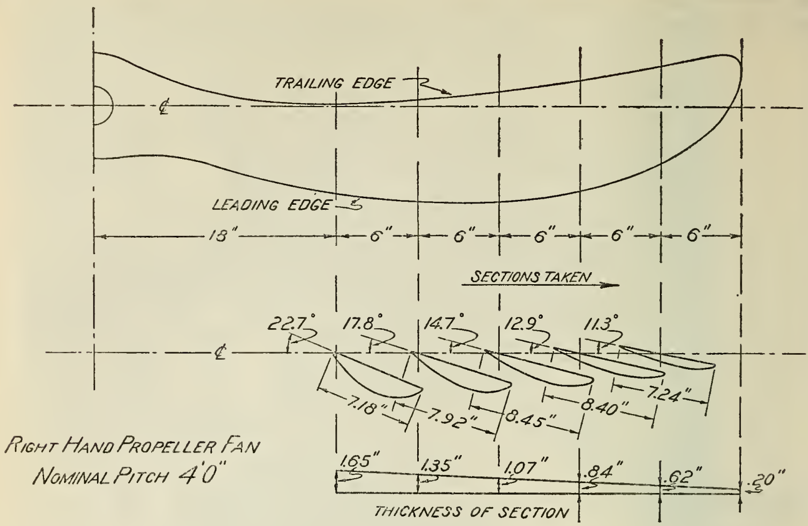

FIGURE 5

hence, it was possible to place the anemometer at any point on the circle swept out when the arm was turned.

The speed of rotation was measured by a Weston electric tachometer directly connected to the motor shaft. The power input was measured

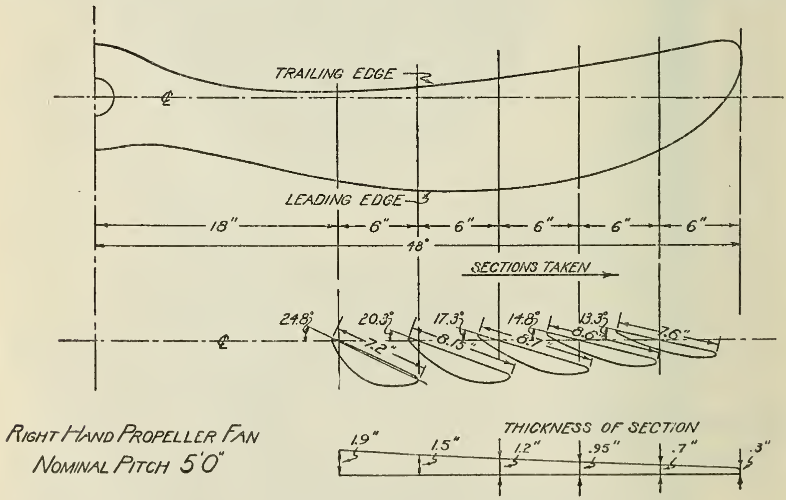

FIGURE 6

by means of a portable-type ammeter and voltmeter accurate to one-half of 1 per cent.

Seven 8-foot 2-blade propeller fans of laminated wood construction were used in the investigation. The nominal pitch of the fans was

2, 3, 4, 5, 6, 7½, and 8½ feet, respectively. The measured geometric pitch at several radii is given in Table 1. Figures 3 to 9 show the approximate shape and dimensions of the fans. It should be noted that the fans are not all of one family, since the plan form, blade width, and blade section vary considerably.

TABLE 1.—*Measured pitch of fans at various radii*

Nominal pitch	Radius in feet				
	1.5	2.0	2.5	3.0	3.5
2-foot	2.14	1.80	1.86	1.80	1.74
3-foot	3.02	2.91	2.98	2.94	2.90
4-foot	3.94	4.03	4.12	4.32	4.40
5-foot	4.40	4.64	4.85	4.97	5.08
6-foot	4.40	5.02	5.44	5.74	5.73
7.5-foot	5.18	6.45	7.32	7.55	7.51
8.5-foot	6.83	7.85	8.33	8.52	8.36

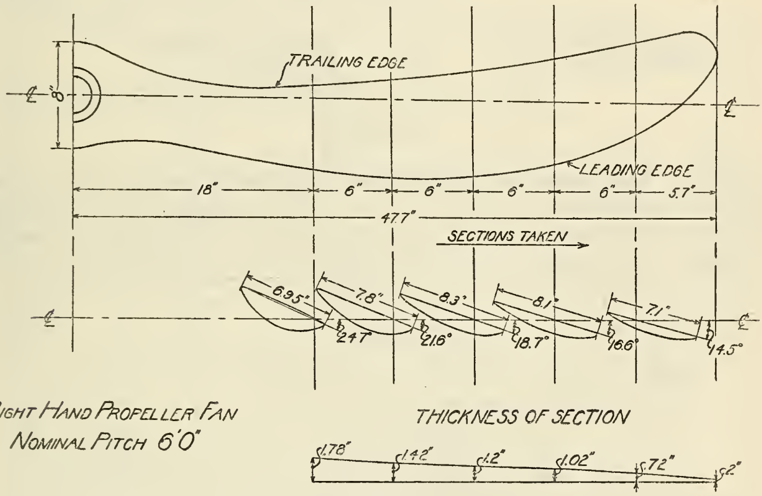

FIGURE 7

The diameter of the duct was made 2 feet larger than the diameter of the fan so as to approximate the condition in a particular cooling-tower installation. That installation has several fans located in the walls of the tower. The cross-sectional area of the tower is 1.56 times the total fan disk area; hence, in reproducing this condition, the ratio of the diameter of the outlet duct to the diameter of the fan was made $\sqrt{1.56}$, or 1.25. A cylindrical ring 4½ inches long and 8 feet 1 inch inside diameter was centered in the plane of the fan to simulate the wall condition.

IV. REMARKS ON THE LOCATION OF THE STATIC PRESSURE HOLES

The static pressure holes were originally located 3⅜ inches downstream from the plane of the fan. With the 5-foot pitch fan operating as a blower at 585 r. p. m., the static pressure was found to be 0.25

inch of water less than the atmospheric pressure. It seemed surprising at first that the fan operating as a blower should develop a decreased static pressure, but further investigation showed it to be a perfectly natural phenomenon, as explained below. A longitudinal

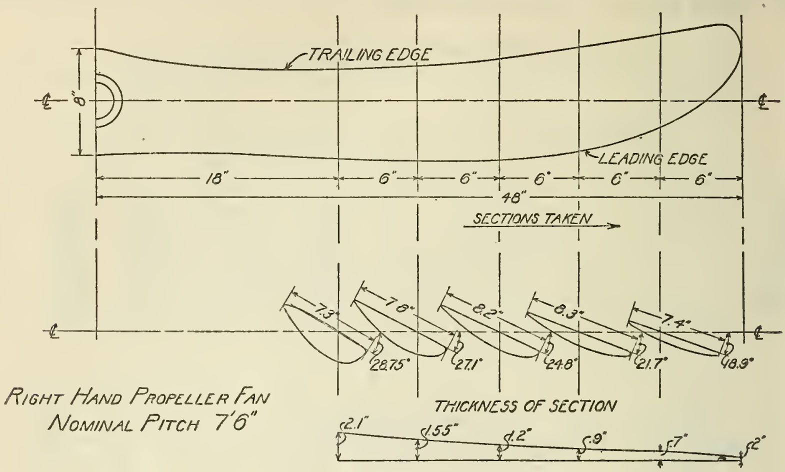

FIGURE 8

traverse of the duct was made, observations of static pressure being taken at the points *1, 2, 3, ... 24* of Figure 10. The results are plotted in Figure 12. The fan produces a condition of reduced pressure on its inlet side which causes the air to flow into the fan.

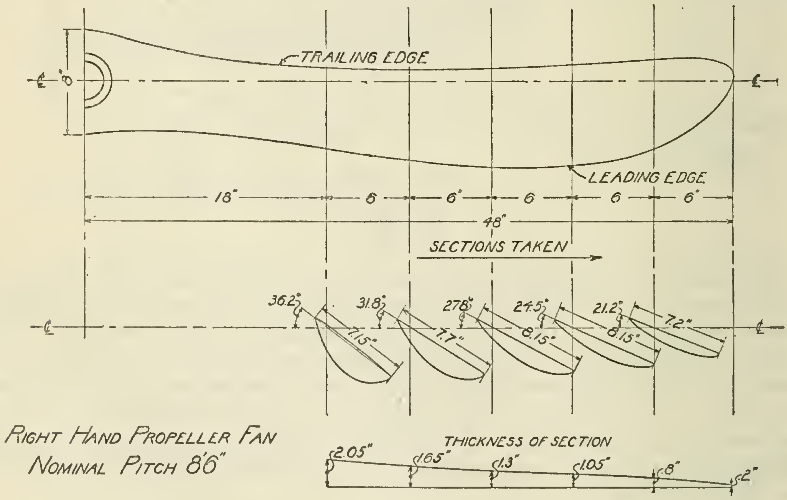

FIGURE 9

Referring to the curve, we see that the static pressure at this section is 0.57 inch of water less than the atmospheric pressure. The fan boosts the pressure to 0.25 inch of water less than the atmospheric pressure and the air is discharged into the duct which is of larger cross section than the inlet ring. The air tends to fill the duct,

slowing up and thus converting velocity pressure into static pressure. The obstruction of the duct by the motor and its supporting columns

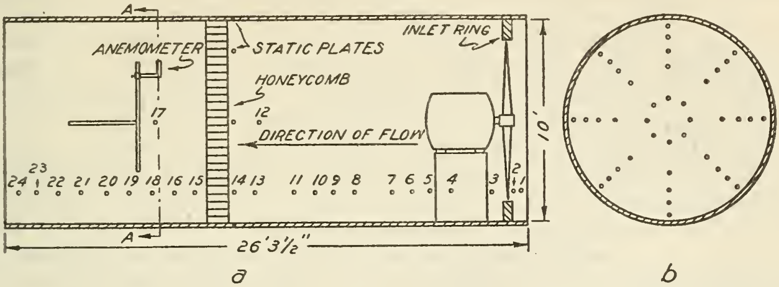

FIGURE 10.—*Arrangement of duct for blower condition*

a, Longitudinal section. *b*, Section *A–A*, showing points of observation for velocity.

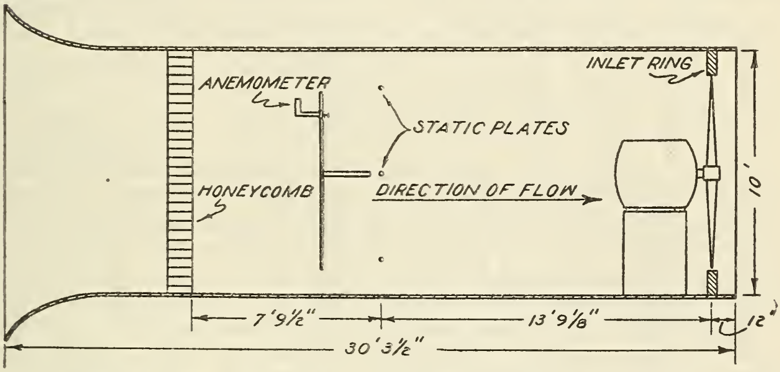

FIGURE 11.—*Longitudinal section of duct, exhaust condition*

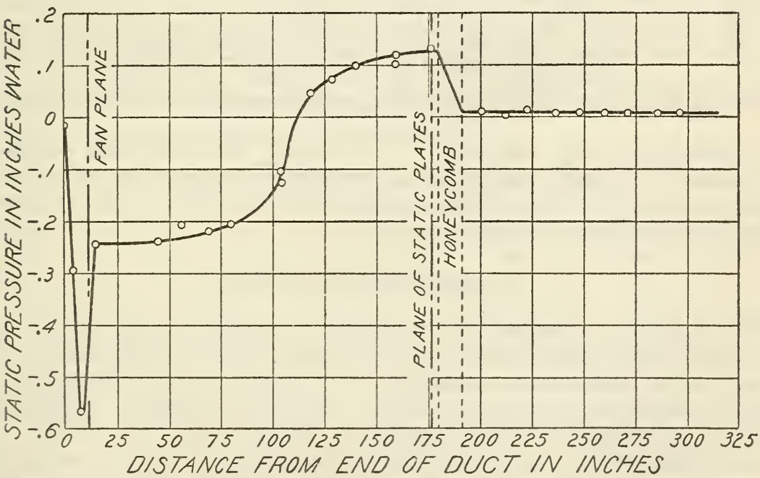

FIGURE 12.—*Results of longitudinal static traverse for the blower condition*

hinders the process until the obstruction has been passed, when we find the pressure rising rapidly to a maximum value just in front of the honeycomb. There is a sharp drop in pressure through the

honeycomb, and a fairly constant value is then maintained to the end of the duct. The pressures measured near the fan are probably greatly in error because of the very turbulent and nonaxial motion of the air. The curve shows clearly that the most desirable location of the static pressure holes is just in front of the honeycomb. Since in the blowing tests some spiral motion persists right up to the honeycomb, we feel that the measurement of the static pressure is less satisfactory in the blowing tests than in the exhaust tests.

The results of the longitudinal traverse for the exhaust condition are shown in Figure 13. In front of the honeycomb the static pressure is about 0.06 inch of water less than the atmospheric pressure. There is a drop through the honeycomb to about 0.1 inch of water less than the atmospheric pressure, and this value is maintained until the obstruction of the passage by the motor and its supports causes the velocity to increase and the pressure to decrease. The fan boosts the pressure to about atmospheric pressure and the air is discharged

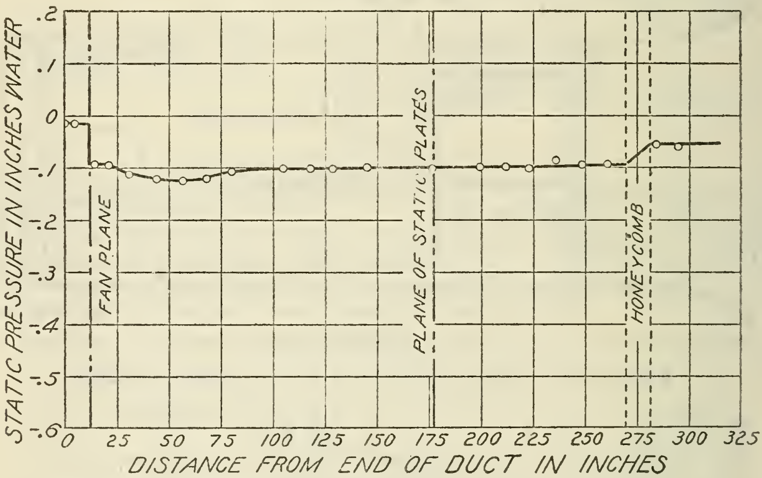

Figure 13.—*Results of the longitudinal static traverse for the exhaust condition*

at a relatively high velocity. It is readily seen that the position of the static holes is not so restricted as for the blower-fan arrangement. The pressures for the exhaust arrangement were very much steadier than for the blower arrangement.

V. TEST PROCEDURE

The duct is not housed in a building, but stands in the open. Runs were not made when there was any precipitation or appreciable wind. The desired resistance conditions were obtained by placing a suitable number of layers of material (mosquito netting, cheese cloth, batiste, or paper) on the upstream side of the honeycomb. The several fans were run at the speeds mentioned above for each resistance condition; the resistance was then changed, and the procedure repeated. Measurements of the velocity for each resistance condition were made at the 40 points indicated in Figure 10 (*b*), the points being at the centers of equal areas of the cross section. Each reading of the anemometer took about one minute and represented an inte-

gration of the velocity over this period of time. The velocity in feet per minute was computed from the time (measured by a stop watch) required for the anemometer to make a counted number of revolutions, the calibration curve being used.

The first set of data was taken with the fan operating as a blower; Figure 10 shows the arrangement in the duct. For the exhaust condition, a faired section 4 feet long and 14 feet maximum diameter (fig. 2) was placed at the opposite end of the duct from the fan so as to secure a more even flow. Also, the honeycomb was moved to the position shown in Figure 11, the anemometer mounting was moved to the opposite side of the honeycomb, the fans were reversed on their hubs, and the direction of rotation of the motor reversed. Observations were taken as described for the blower fan.

At the beginning of the test the efficiency of the motor at the various speeds was obtained by means of a Prony brake. Curves of the efficiency against power input were plotted and used to compute the power delivered to the fan from the observed power input to the motor. Check runs made at the end of the tests showed an apparent change of about 5 per cent in the power absorbed by the fans. Check measurements of the pitch showed no apparent change in the fans, and, therefore, the Prony brake was again rigged up, and several efficiency runs were made. The results showed that the efficiency of the motor had changed during the tests. Since there was no way of knowing when the change took place, all of the fans were run again at the various speeds and against the various resistance conditions. Observations were made as before, except that the velocity readings were omitted. This omission was justified because the head-volume relation was not affected by the change in the efficiency of the motor. In plotting the power-volume curve, the head-volume curve obtained in the original tests was used to obtain the volume corresponding to the observed head. To guard against errors due to further changes in the efficiency of the motor, measurements of the power required to turn the motor without fan load were made at frequent intervals, and since the efficiencies measured by the Prony brake were in good agreement with those determined from the no-load power and the electrical losses, the power absorbed by the fan was computed by subtracting the electrical and mechanical losses.

VI. REDUCTION OF OBSERVATIONS

When a fan is operated at various rotational speeds to move air through a given duct system, the volume of air moved varies directly as the speed of rotation N; the static pressure developed varies as the square of the speed of rotation, and the power required varies as the cube of the speed of rotation. The static pressure and the power also vary directly as the air density, if the volume transferred per minute is the same in all cases, as may be secured by suitably adjusting the speed of rotation. For geometrically similar installations, the average velocity of the moving air is proportional to the tip speed, hence to the product of speed of rotation (N) and diameter (D). The volume, therefore, varies as ND^3 (air speed times area of duct), the head as N^2D^2 (square of the tip speed), and the power as N^3D^5 (product of head and volume). It is assumed that the reader

is familiar with these relations, which form the basis of practically all fan tables.

The resistance of a given duct system is not constant, but is a function of the air speed, varying approximately as the square of the volume of air flowing. When the speed of rotation of a fan delivering air to the system is increased, the volume increases directly as the speed of rotation, and the pressure developed by the fan increases as the square of the speed of rotation, hence, as the square of the volume flowing. The pressure developed by the fan keeps in step with the increasing resistance of the duct and, therefore, except for the small and usually negligible deviations from the relations stated, no additional information is obtained by changing merely the speed of rotation of the fan that is delivering air to, or removing it from, a fixed duct system.

A fixed value of the ratio of the resistance of the duct to the square of the volume flowing may be spoken of as a given resistance ratio. To make a satisfactory test of the fan, this ratio must be varied from infinity (a closed duct) to nearly zero (an open duct). The method of doing this has already been explained. With a closed duct, the fan builds up a static pressure in the duct, but there is no flow of air through the duct system. As the resistance to flow is decreased, the speed of rotation of the fan remaining constant, the volume of air flowing increases and the static pressure decreases until finally, when the difference in static pressure between the inlet and outlet side of the fan is zero, the fan moves the greatest volume of air. Curves of the static pressure and power against air volume for a constant speed of rotation represent one method of giving the characteristics of the fan.

The useful work of the fan is usually defined as the work done in moving the observed volume of air against the observed static pressure. With this definition the efficiency of a fan operating either in a closed duct or in the open is zero. Often, but not always, the work required to impart the observed velocity to the air is included in the useful work. We have included it. A volume, Q cu. ft./min., moving in a duct of diameter, d feet, corresponds to a mean speed of $\frac{Q}{0.7854\ d^2 \times 60}$ ft./sec. The kinetic energy imparted per second is one-half the product of the mass of air moved per second times the square of the mean speed. The mass moved per second is the product of the density (ρ slugs/ft.^{3}) by the volume per second $\left(\frac{Q}{60}\text{ cu. ft.}\right)$. Hence the kinetic energy imparted per second is $\frac{1}{2}\rho\frac{Q}{60}\left(\frac{Q}{0.7854\ d^2\times 60}\right)^2$ ft. lbs./sec. The power required to move the volume $\frac{Q}{60}$ cu. ft./sec. against a static pressure of h inches of water ($=5.2\ h$ lbs./ft.^{2}) is $\frac{5.2\ Qh}{60}$ ft. lbs./sec. Hence, the total useful power is $\frac{Q}{60}\left\{5.2h+\frac{1}{2}\rho\left(\frac{Q}{0.7854\ d^2\times 60}\right)^2\right\}$ ft. lbs./sec. The second term in the bracket is the velocity pressure, and the sum of the two terms is the total pressure, both expressed in lbs./ft.^{2} We have adopted, then, as the useful

power the power required to move the observed volume from the total pressure on the inlet side of the fan to the total pressure on the outlet side.

We do not intend to suggest that the above definition of useful work is the "true" or "best." The practical problem is generally to circulate a definite quantity of air through a given duct system with the minimum power. The use of the conception of efficiency may or may not be helpful in securing the desired result.

All observations were reduced to the standard air conditions adopted by the Fan Manufacturers Association,[1] that is, to air weighing 0.07488 lbs./ft.^{3}. This weight corresponds to air having a barometric pressure of 29.92 inches of mercury, standard gravity, a dry-bulb temperature of 68° F., and 50 per cent relative humidity.

In view of the approximate laws previously stated and to facilitate the use of the results in estimating the performance of fans of other diameters and rotational speeds, the results are given in terms of the coefficients defined as follows:

$$\left.\begin{aligned} K_H &= 10^8\frac{H}{N^2D^2} \\ K_Q &= \frac{Q}{ND^3} \\ K_P &= 10^{12}\frac{P}{N^3D^5} \end{aligned}\right\} \qquad (1)$$

$$\text{Efficiency} = \frac{5.2\ QH}{33{,}000\ P} = \frac{10^4 \times 5.2\ K_HK_Q}{33{,}000\ K_P}$$

where, K_H = total pressure coefficient, K_Q = volume coefficient, K_P = power coefficient, N = rotational speed (r. p. m.), D = diameter of fan (ft.), Q = volume of air (cu. ft./min.), P = horsepower absorbed by fan, and H = increase in total pressure produced by fan (inches of water), that is, the sum of the increase in the static pressure plus the increase (or minus the decrease) in the velocity pressure.

The increase in static pressure is measured on the manometer gage. The velocity pressure in a duct of diameter, d, is computed from the expression already given,

$$\frac{\frac{1}{2}\rho}{5.2}\left(\frac{Q}{0.7854\ d^2 \times 60}\right)^2$$

inches of water. The standard value of ρ is $\frac{0.07488}{32.156} = 0.002329$ slugs/ft.^{3}. Combining the numerical factors, we find that the velocity pressure is

$$0.0000001008\frac{Q^2}{d^4} = \frac{N^2D^2}{10^8}\left(10.08\ K_Q{}^2\frac{D^4}{d^4}\right)$$

inches of water, ρ having its standard value.

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[1] J. A. S. H. V. E., **29**, p. 371; 1923.

For the blower condition, the air starts from rest at atmospheric pressure, and H is the excess of the total pressure in a 10-foot duct above that pressure. As D is 8 feet, the increase in velocity pressure is $\frac{N^2D^2}{10^8}(4.13\ K_Q^2)$, and $H = H_S + \frac{N^2D^2}{10^8}(4.13\ K_Q^2)$, H_S being the excess of the static pressure in the duct above one atmosphere. Hence, $K_H = \frac{10^8 H_S}{N^2D^2} + 4.13\ K_Q^2$.

For the exhaust condition, D being 8 feet, the increase in the velocity pressure in passing from the 10-foot duct, where the static pressure is H_S below atmospheric, to the outside air (atmospheric pressure) at an opening 8 feet 1 inch in diameter will be $\frac{N^2D^2}{10^8}$ $(10.08\ K_Q^2D^4)\left\{\frac{1}{(8.083)^4} - \frac{1}{10^4}\right\} = \frac{N^2D^2}{10^8}(5.55\ K_Q^2)$, provided that the average air speed is the same at every point of the 8 foot 1 inch opening. Then $H = H_S + \frac{N^2D^2}{10^8}(5.55\ K_Q^2)$, and $K_H = \frac{10^8 H_S}{N^2D^2} + 5.55\ K_Q^2$. (Actually, the flow will not be distributed uniformly over the discharge opening, and this will modify the results.)

When the coefficients, K_H, K_Q, and K_P are known, it is easy to determine whether given conditions can be met, and if so, what rotational speed, diameter, and pitch are required. This is facilitated by combining these equations in various ways and by the use of suitable graphs. For example—

$$D^4 = \frac{Q^2 K_H}{10^8 H K_Q^2} \tag{2}$$

$$N = \frac{Q}{D^3 K_Q}$$

are convenient for use with a slide rule. These two equations may be used in the solution of many types of selection problems. For special problems it may be convenient to plot the characteristic curves in terms of new variables so chosen that the independent variable involves only known quantities. For example if Q, N, and H are the known quantities, some convenient power of $\frac{K_Q^2}{K_H^3}$ is chosen as the independent variable, for by the relations given in equation (1)

$$\frac{K_Q^2}{K_H^3} = \frac{Q^2 N^4}{10^{24} H^3}$$

and, hence, its value is determined completely by the known quantities Q, N, and H. The reader is referred to a paper entitled "Characteristics of Centrifugal Fans," by T. G. Estep and C. A. Carpenter, appearing in the Proceedings of the Engineering Society of Western Pennsylvania, volume 43, 1927–28, page 306, for illustrations of the use of this independent variable.

If Q, D, and H are the known quantities, a convenient power of $\frac{K_Q^2}{K_H}$ is chosen as the independent variable, for by the relations given in equation (1)

$$\frac{K_Q^2}{K_H}=\frac{Q^2}{10^8 D^4 H}\cdot$$

and, hence, its value is determined completely by Q, D, and H. This variable forms the basis of the so-called Hagen chart described in Marks Mechanical Engineers Handbook, second edition, page 1643 (McGraw-Hill, 1924).

TABLE 2.—*Blower fan*

(*a*) 2-FOOT PITCH (PITCH/DIAMETER RATIO 0.250)

800 r. p. m.				900 r. p. m.			
K_H	K_Q	K_P[1]	Efficiency	K_H	K_Q	K_P[1]	Efficiency
			Per cent				*Per cent*
0.245	0.1870	0.33	21.9	0.233	0.1848	0.33	20.6
.511	.1377	.34	32.6	.506	.1375	.34	32.2
.662	.1123	.33	35.5	.674	.1128	.33	36.3
.938	.0672	.29	34.2	.945	.0676	.29	34.7
1.075	.0346	.24	24.4	1.065	.0346	.24	24.2
1.233	0	.21	0	1.224	0	.21	0

(*b*) 3-FOOT PITCH (PITCH/DIAMETER RATIO 0.375)

0.404	0.2447	0.58	26.8	0.414	0.2410	0.58	27.1
.815	.1743	.58	38.6	.815	.1757	.58	38.9
1.073	.1399	.57	41.5	1.062	.1375	.56	41.1
1.362	.0811	.48	36.2	1.352	.0701	.46	32.5
1.449	.0463	.42	25.2	1.467	.0469	.42	25.8
1.695	0	.34	0	1.700	0	.34	0

(*c*) 4-FOOT PITCH (PITCH/DIAMETER RATIO 0.500)

585 r. p. m.				700 r. p. m.			
K_H	K_Q	K_P[1]	Efficiency	K_H	K_Q	K_P[1]	Efficiency
			Per cent				*Per cent*
0.580	0.2740	0.81	30.9	0.563	0.2775	0.81	30.4
1.089	.2000	.81	42.4	1.060	.2005	.81	41.4
1.362	.1552	.76	43.8	1.360	.1575	.77	44.0
1.568	.1015	.69	36.3	1.551	.1051	.70	36.7
1.855	.0420	.60	20.5	1.854	.0442	.60	21.5
2.031	0	.52	0	2.012	0	.52	0

(*d*) 5-FOOT PITCH (PITCH/DIAMETER RATIO 0.625)

0.659	0.297	1.04	29.6	0.655	0.299	1.04	29.7
.968	.258	1.03	38.2	.951	.256	1.03	37.3
1.277	.208	1.01	41.5	1.275	.209	1.01	41.6
1.528	.129	.93	33.4	1.497	.135	.94	33.9
1.800	.070	.81	24.5	1.780	.074	.81	25.6
2.110	0	.68	0	2.140	0	.68	0

[1] These values of K_P taken from faired curves.

TABLE 2.—*Blower fan*—Continued

(*e*) 6-FOOT PITCH (PITCH/DIAMETER RATIO 0.750)

585 r. p. m.				700 r. p. m.			
K_H	K_Q	K_P [1]	Efficiency	K_H	K_Q	K_P [1]	Efficiency
			Per cent				*Per cent*
0.712	0.3185	1.15	31.1	0.683	0.3090	1.14	29.2
.714	.3068	1.14	30.3	.695	.3075	1.14	29.5
1.288	.2151	1.06	41.2	1.313	.2161	1.06	42.2
1.033	.2597	1.11	38.1	1.048	.2577	1.10	38.7
1.521	.1529	.98	37.4	1.508	.1562	.98	37.9
1.343	.2055	1.05	41.4	1.345	.2125	1.06	42.5
1.611	.0992	.92	27.4	1.645	.1006	.92	28.3
1.516	.1332	.96	33.2	1.515	.1361	.96	33.9
1.988	.0449	.83	16.9	1.981	.0475	.84	17.6
1.837	.0621	.86	20.9	1.871	.0596	.86	20.4
2.100	0	.75	0	2.108	0	.75	0
2.062	0	.75	0	2.080	0	.75	0

(*f*) 7.5-FOOT PITCH (PITCH/DIAMETER RATIO 0.938)

0.949	0.3525	1.60	32.9	0.876	0.3505	1.60	30.2
1.290	.2923	1.60	37.1	1.314	.3014	1.60	39.0
1.830	.2250	1.60	40.5	1.847	.2220	1.60	40.4
1.928	.1499	1.50	30.4	1.947	.1505	1.50	30.8
2.357	.0972	1.39	25.9	2.285	.0988	1.39	25.6
2.673	0	1.31	0	2.695	0	1.31	0

(*g*) 8.5-FOOT PITCH (PITCH/DIAMETER RATIO 1.063)

----------	----------	----------	----------	0.977	0.3740	1.99	28.9
0.992	0.3670	1.98	29.0	.988	.3660	1.98	28.8
1.332	.3068	1.94	33.2	1.355	.3058	1.94	33.6
1.696	.2423	1.88	34.5	1.704	.2368	1.88	33.8
2.003	.1486	1.68	27.9	1.949	.1505	1.68	27.5
2.019	.1455	1.68	27.5	1.951	.1524	1.68	27.9
2.055	.1418	1.67	27.5	2.041	.1396	1.66	27.1
2.384	.0901	1.64	20.6	2.366	.0960	1.64	21.8
2.629	0	1.59	0	2.610	0	1.59	0

TABLE 3.—*Exhaust fan*

(*a*) 2-FOOT PITCH (PITCH/DIAMETER RATIO 0.250)

800 r. p. m.				900 r. p. m.			
K_H	K_Q	K_P [1]	Efficiency	K_H	K_Q	K_P [1]	Efficiency
			Per cent				*Per cent*
0.338	0.1561	0.34	24.5	0.325	0.1555	0.34	23.4
.538	.1234	.33	31.7	.523	.1236	.33	30.9
.640	.0997	.32	31.4	.644	.1004	.32	31.8
.858	.0688	.30	31.0	.859	.0697	.30	31.4
.968	.0509	.28	27.7	.969	.0524	.28	28.6
1.165	0	.22	0	1.165	0	.22	0

(*b*) 3-FOOT PITCH (PITCH/DIAMETER RATIO 0.375)

0.522	0.1978	0.58	28.0	0.512	0.1970	0.58	27.4
.818	.1554	.57	35.1	.807	.1546	.57	34.5
.953	.1282	.55	35.0	.952	.1284	.55	35.0
1.181	.0924	.50	34.4	1.191	.0901	.50	33.8
1.391	.0632	.47	29.5	1.375	.0648	.47	29.9
1.614	0	.36	0	1.612	0	.36	0

[1] These values of K_P taken from faired curves.

TABLE 3.—*Exhaust fan*—Continued

(*c*) 4-FOOT PITCH (PITCH/DIAMETER RATIO 0.500)

585 r. p. m.				700 r. p. m.			
K_H	K_Q	K_P[1]	Efficiency	K_H	K_Q	K_P[1]	Efficiency
			Per cent				*Per cent*
0.683	0.2277	0.81	30.2	0.660	0.2228	0.81	28.6
.690	.2267	.81	30.4	.687	.2268	.81	30.3
1.036	.1772	.80	36.1	1.023	.1772	.80	35.7
1.245	.1347	.77	34.3	1.231	.1391	.78	34.6
1.473	.1024	.75	31.7	1.426	.1038	.75	31.1
1.648	.0616	.69	23.2	1.651	.0651	.70	24.2
1.927	0	.55	0	1.912	0	.55	0

(*d*) 5-FOOT PITCH (PITCH/DIAMETER RATIO 0.625)

0.778	0.2388	1.03	28.4	0.784	0.2399	1.03	28.8
.939	.2138	1.02	31.0	.926	.2192	1.02	31.4
1.090	.1861	1.00	32.0	1.072	.1878	1.00	31.7
1.280	.1533	.98	31.5	1.260	.1544	.98	31.3
				1.584	.1071	.91	29.4
1.563	.1060	.91	28.7	1.538	.1107	.92	29.2
1.753	.0717	.84	23.6	1.747	.0752	.85	24.3
2.040	0	.69	0	2.030	0	.69	0

(*e*) 6-FOOT PITCH (PITCH/DIAMETER RATIO 0.750)

0.756	0.2353	1.08	25.9	0.751	0.2363	1.08	25.9
.891	.2121	1.07	27.8	.897	.2148	1.07	28.4
1.037	.1816	1.05	28.3	1.030	.1828	1.05	28.3
1.171	.1506	1.03	27.0	1.184	.1517	1.03	27.4
1.518	.1036	.97	25.6	1.507	.1075	.97	26.3
1.713	.0696	.91	20.6	1.695	.0748	.92	21.7
1.950	0	.81	0	1.932	0	.81	0

(*f*) 7.5-FOOT PITCH (PITCH/DIAMETER RATIO 0.938)

1.058	0.2797	1.60	29.1	1.066	0.2816	1.60	29.6
1.216	.2507	1.63	29.5	1.212	.2508	1.63	29.4
1.337	.2106	1.65	26.9	1.328	.2123	1.65	26.9
1.577	.1683	1.61	26.0	1.572	.1720	1.62	26.3
1.976	.1292	1.51	26.6	1.949	.1307	1.52	26.4
2.210	.0801	1.40	19.9	2.160	.0844	1.40	20.5
2.392	0	1.30	0	2.380	0	1.30	0

(*g*) 8.5-FOOT PITCH (PITCH/DIAMETER RATIO 1.063)

1.110	0.2865	1.93	25.9	1.088	0.2845	1.93	25.3
1.264	.2552	1.96	25.9	1.246	.2541	1.96	25.5
1.384	.2190	1.98	24.1	1.375	.2181	1.98	23.9
1.703	.1743	1.85	25.3	1.668	.1770	1.86	25.0
2.018	.1359	1.72	25.1	1.978	.1378	1.72	25.0
2.129	.0758	1.64	15.5				
2.154	.0831	1.65	17.1	2.116	.0869	1.65	17.5
2.269	0	1.59	0	2.238	0	1.59	0

[1] These values of K_P taken from faired curves.

Each of these independent variables has advantages in a particular type of problem, eliminating methods of trial and error which may be necessary with the characteristic curve involving K_H and K_Q. To avoid possible confusion due to a number of methods of procedure, we will make use of the relations of equations (2) and the K_H–K_Q curves in the illustrative problems given in this paper.

VII. RESULTS

The results obtained are summarized in Tables 2, (*a*) to (*g*), and 3 (*a*) to (*g*), and in Figures 14 to 31. As the pitch/diameter ratio is increased, the curve of efficiency *v.* volume coefficient becomes pro-

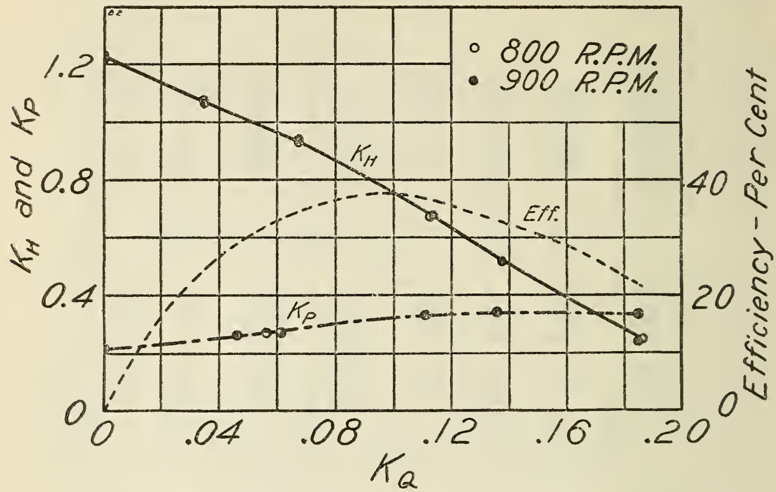

FIGURE 14.—*Characteristics of 8-foot by 2-foot blower fan*

Diameter, 8 feet; pitch, 2 feet; pitch/diameter ratio—0.250.

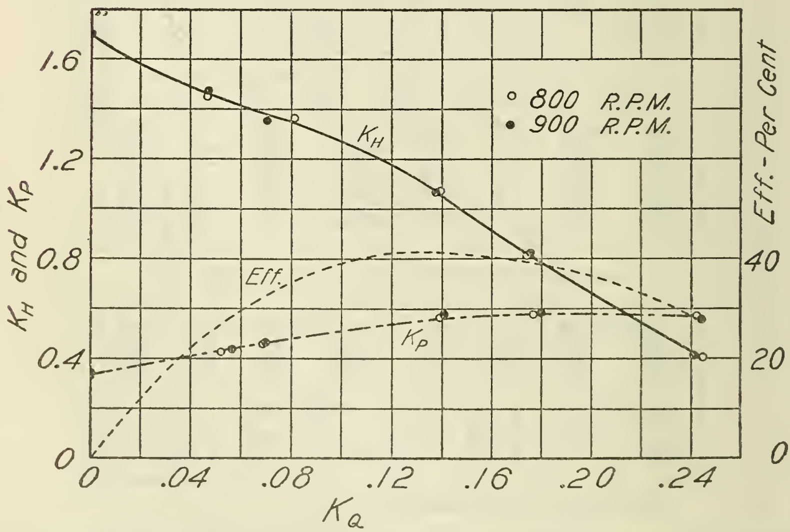

FIGURE 15.—*Characteristics of 8-foot by 3-foot blower fan*

Diameter, 8 feet; pitch, 3 feet; pitch/diameter ratio—0.375.

gressively flatter (figs. 29 and 31), and the useful values of the coefficient of volume and of head progressively increase up to a pitch/diameter

ratio of 0.938.[2] Hence, with the higher ratios larger values of the coefficients are available, and the diameter and the speed of rotation

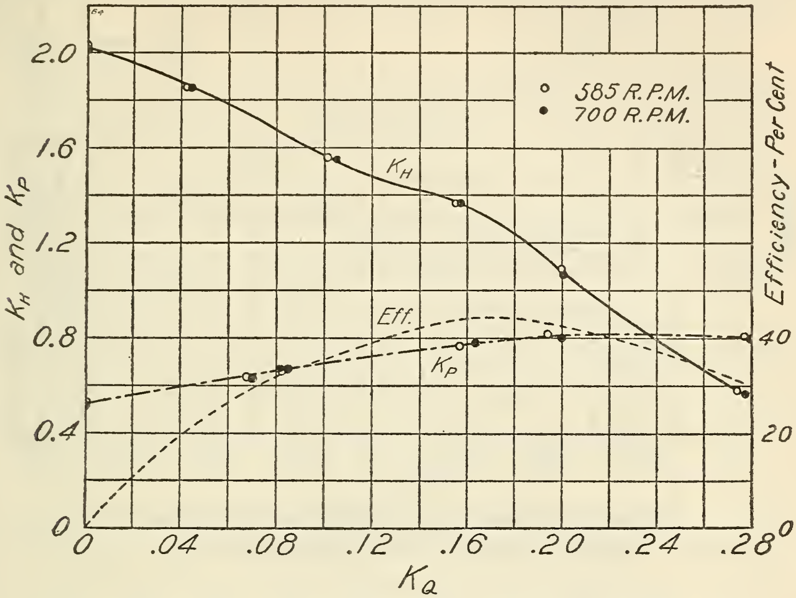

FIGURE 16.—*Characteristics of 8-foot by 4-foot blower fan*
Diameter, 8 feet; pitch, 4 feet; pitch/diameter ratio—0.500.

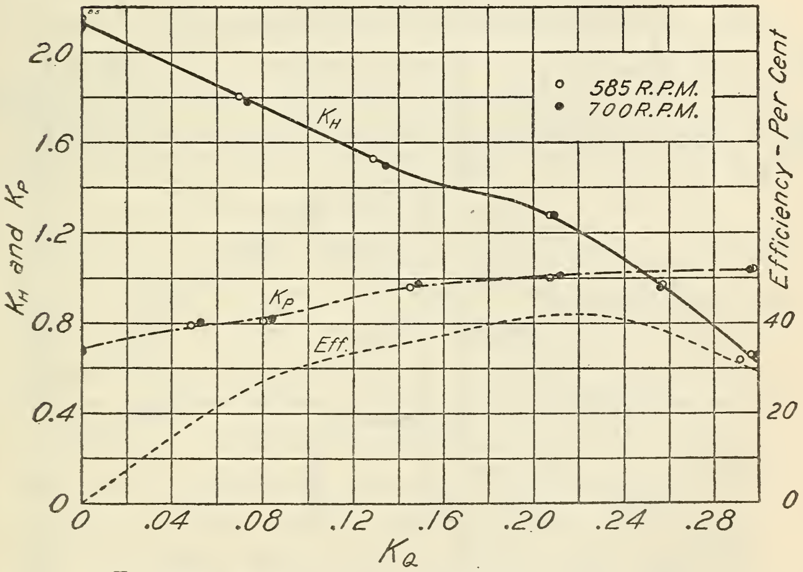

FIGURE 17.—*Characteristics of 8-foot by 5-foot blower fan*
Diameter, 8 feet; pitch, 5 feet; pitch/diameter ratio—0.625.

may be varied, so as to suit given conditions, over a limited range without serious loss in efficiency. The maximum efficiency varies slowly

[2] The coefficients for the fan of pitch/diameter ratio 0.750 are somewhat out of line, probably because the blade widths are smaller than those of the neighboring fans, and the actual pitch is less than the nominal pitch.

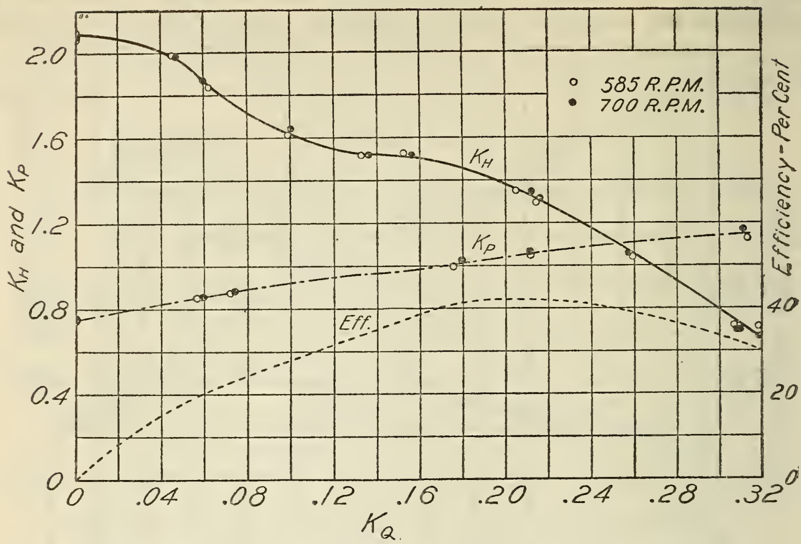

FIGURE 18.—*Characteristics of 8-foot by 6-foot blower fan*

Diameter, 8 feet; pitch, 6 feet; pitch/diameter ratio—0.750.

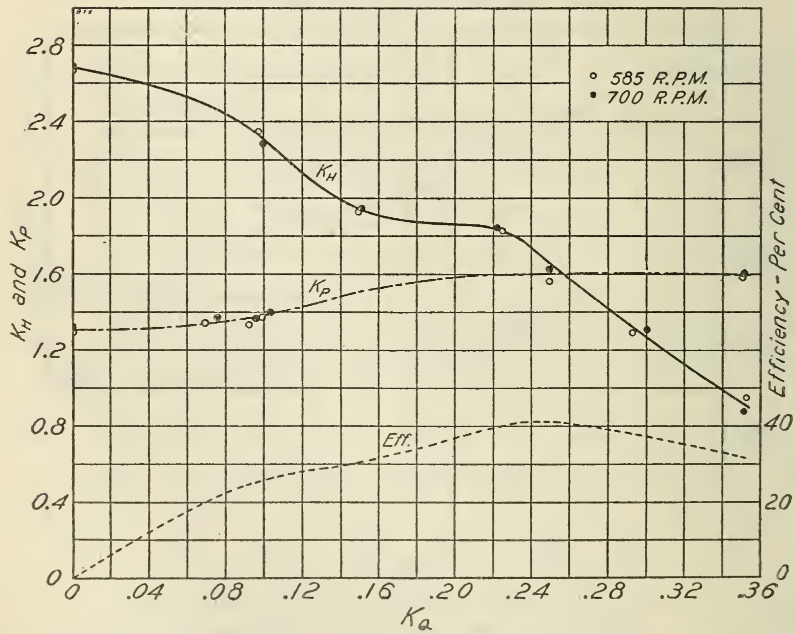

FIGURE 19.—*Characteristics of 8-foot by 7-foot 6-inch blower fan*

Diameter, 8 feet; pitch, 7 feet 6 inches; pitch/diameter ratio—0.938.

with the pitch/diameter ratio, passing through a maximum near the ratio 0.500, and at ratio 1.063 reaching the lowest value observed in this work. In general, ratios less than 0.3 or greater than 1.0 should not be used.

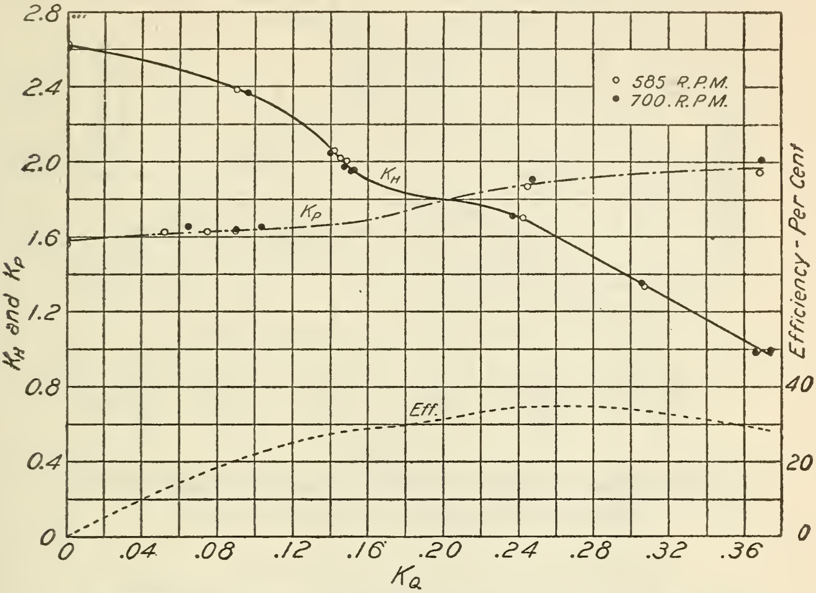

FIGURE 20.—*Characteristics of 8-foot by 8-foot 6-inch blower fan*

Diameter, 8 feet; pitch, 8 feet 6 inches; pitch/diameter ratio—1.063.

In each case, the efficiency of the fan when used for blowing slightly exceeds that when used for exhausting.

To illustrate the method of using the characteristic curves, we may consider some practical problems. It is desired to deliver 34,200

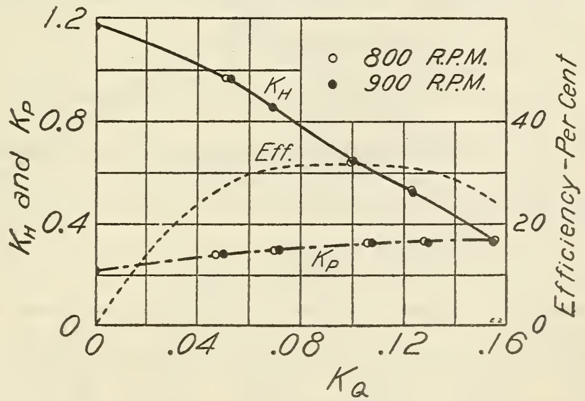

FIGURE 21.—*Characteristics of 8-foot by 2-foot exhaust fan*

Diameter, 8 feet; pitch, 2 feet; pitch/diameter ratio—0.250.

cu. ft./min. against a total pressure of one-half inch of water, using a 2-blade propeller fan 7 feet in diameter and rotating at 700 r. p. m. as a blower. Can this be done; and if so, what will be the pitch of the fan and the power required?

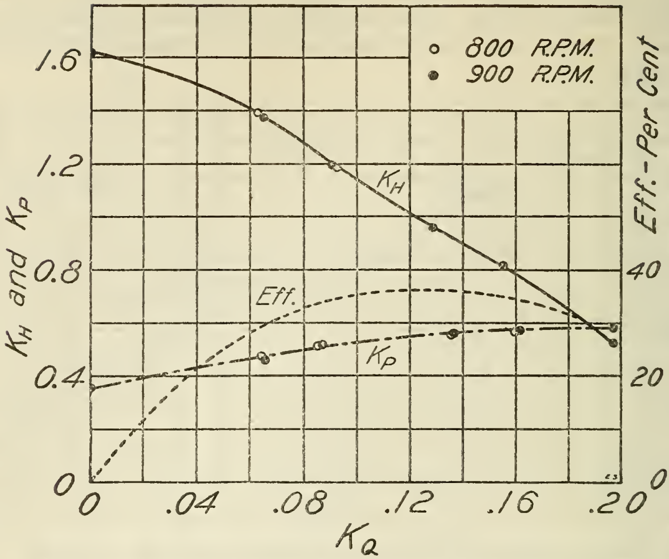

FIGURE 22.—*Characteristics of 8-foot by 3-foot exhaust fan*

Diameter, 8 feet; pitch, 3 feet; pitch/diameter ratio—0.375.

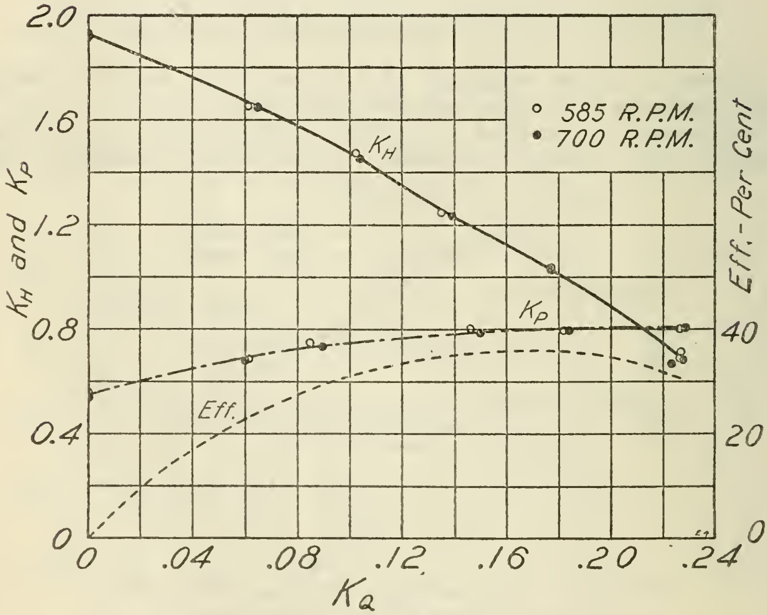

FIGURE 23.—*Characteristics of 8-foot by 4-foot exhaust fan*

Diameter, 8 feet; pitch, 4 feet; pitch/diameter ratio—0.500.

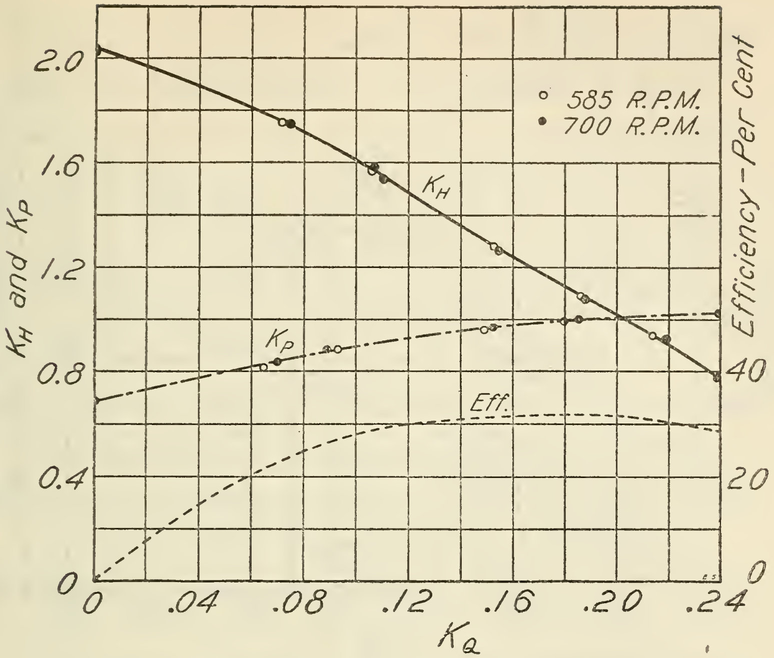

FIGURE 24.—*Characteristics of 8-foot by 5-foot exhaust fan*

Diameter, 8 feet; pitch, 5 feet; pitch/diameter ratio—0.625.

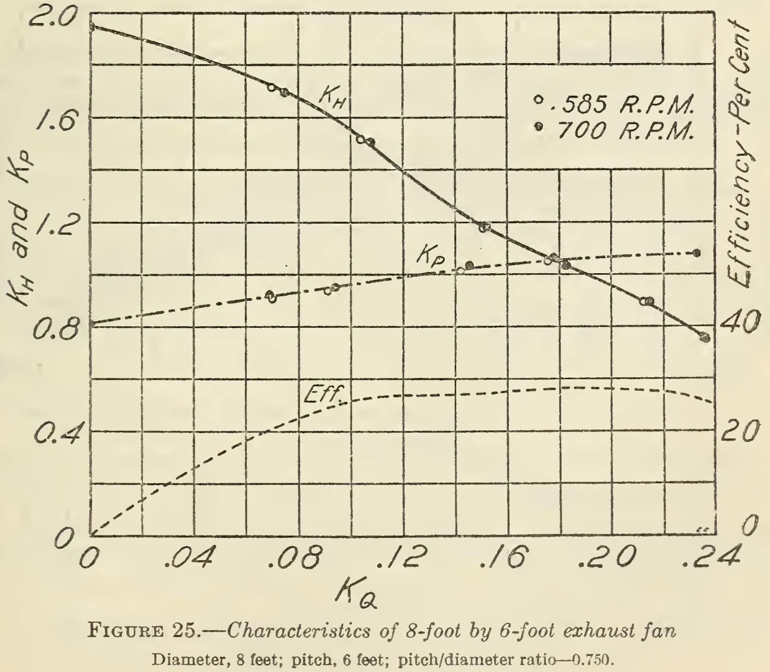

FIGURE 25.—*Characteristics of 8-foot by 6-foot exhaust fan*

Diameter, 8 feet; pitch, 6 feet; pitch/diameter ratio—0.750.

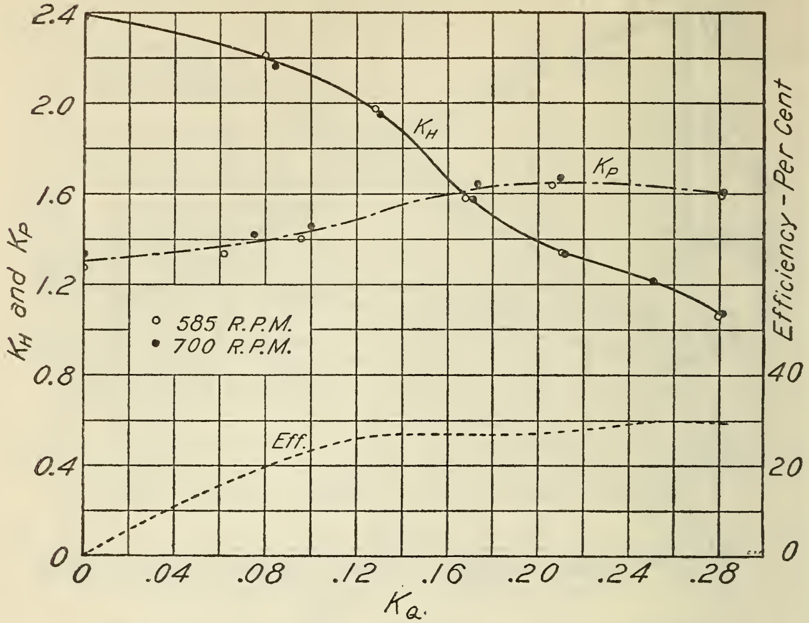

FIGURE 26.—*Characteristics of 8-foot by 7-foot 6-inch exhaust fan*

Diameter, 8 feet; pitch, 7 feet 6 inches; pitch/diameter ratio—0.938.

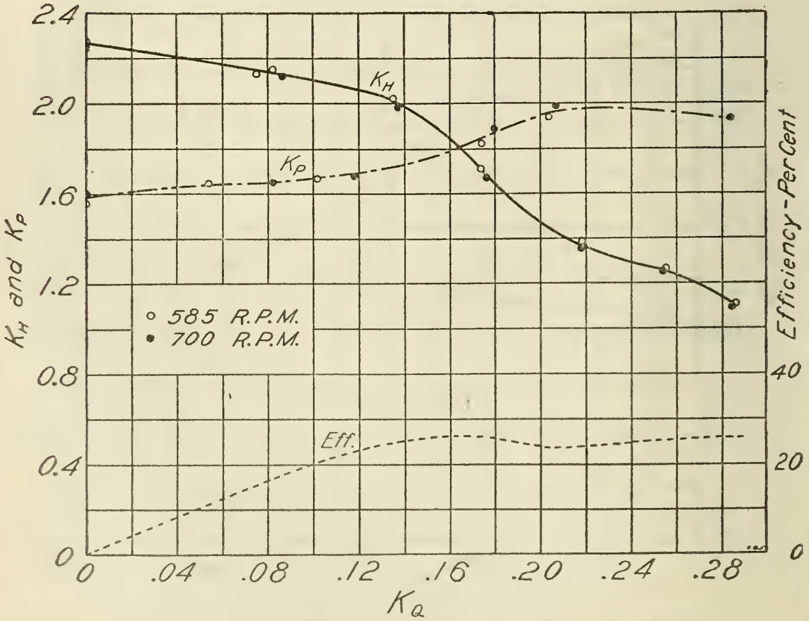

FIGURE 27.—*Characteristics of 8-foot by 8-foot 6-inch exhaust fan*

Diameter, 8 feet; pitch, 8 feet 6 inches; pitch/diameter ratio—1.063.

We find at once that the required $K_Q=\frac{34,200}{700\times 7^3}=0.1424$ and the required $K_H=\frac{10^8\times 0.5}{700^2\times 7^2}=2.082$. An examination of Figures 19 and

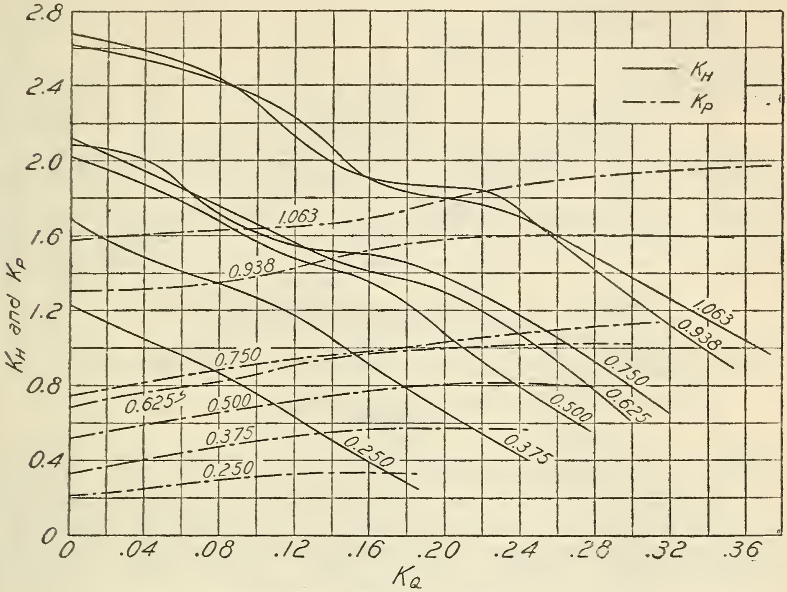

FIGURE 28.—*Characteristics of 2-blade propeller fans of various pitch/diameter ratios*

Blowing condition.

20 shows that this point is above all of the characteristic curves and, therefore, that the desired performance can not be obtained. The nearest point on the characteristic curve for the propeller of

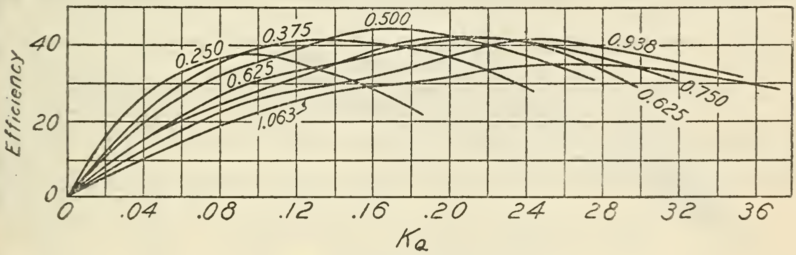

FIGURE 29.—*Efficiency of 2-blade propeller fans of various pitch/diameter ratios*

Blowing condition.

pitch/diameter ratio 0.938 (fig. 19) which has the same value of $\frac{H}{Q^2}$; that is, the same duct resistance, is $K_H=2.000$, $K_Q=0.1395$,[3] corresponding

[3] These values are found by trial and error. From the definitions of K_H and K_Q, $\frac{H}{Q^2}=\frac{10^{-8}}{D^4}\frac{K_H}{K_Q^2}$. But $\frac{H}{Q^2}$ for the desired condition is $\frac{0.5}{34,200^2}$ and D is 7 feet. By trial we look for values of K_H and K_Q on the curve such that $\frac{K_H}{K_Q^2}$ is equal to $\frac{10^8\times 0.5\times 7^4}{34,200^2}=103$. We find the values given.

to a reduction of the volume to 33,500 cu. ft./min. The value of K_P is 1.47 and, hence, the power required is $1.47 \times 10^{-12} \times 700^3 \times 7^5 = 8.48$ h. p. The pitch is of course $0.938 \times 7 = 6.56$ feet.

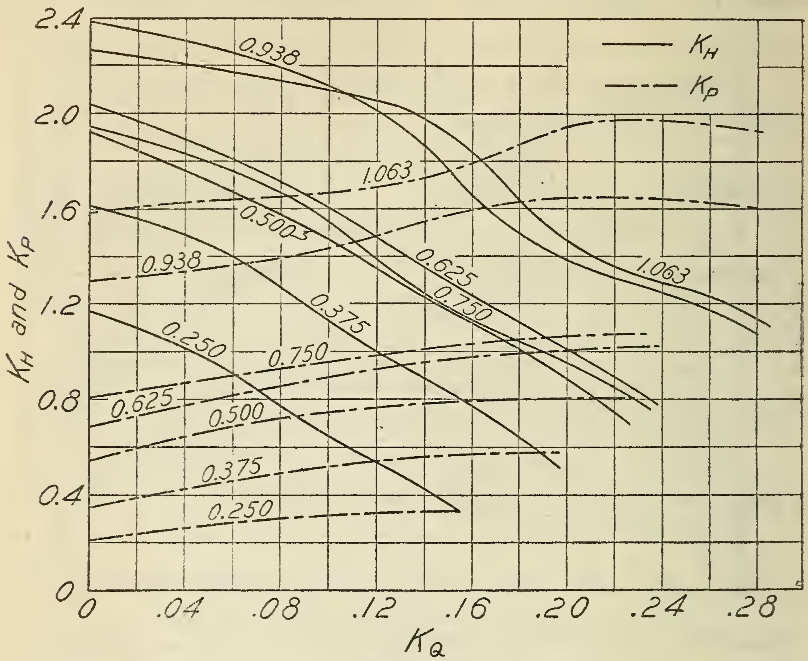

FIGURE 30.—*Characteristics of 2-blade propeller fans of various pitch/diameter ratios*

Exhausting condition.

To secure the desired performance it is necessary to change the diameter or speed of rotation. Suppose we consider a fan of pitch/diameter ratio 0.625 under conditions of maximum efficiency for

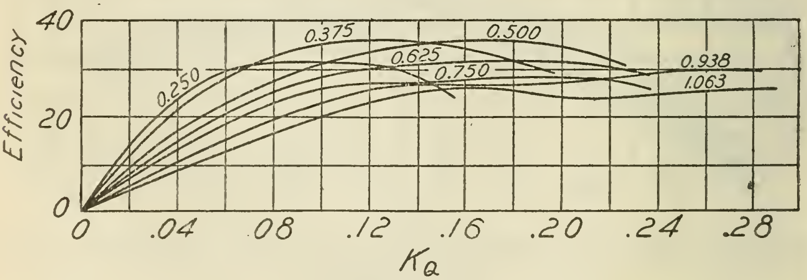

FIGURE 31.—*Efficiency of 2-blade propeller fans of various pitch/diameter ratios*

Exhausting condition.

which $K_H = 1.2$, $K_Q = 0.22$. Hence, by equations (2)

$$D^4 = \frac{1.2 \times 34{,}200^2}{10^8 \times 0.22^2 \times 0.5} = 580$$

whence

$$D = 4.907 \text{ feet.}$$

and

$$N=\frac{34{,}200}{0.22\times 4.907^3}=1{,}314 \text{ r. p. m.}$$

The pitch is $4.907\times 0.625=3.066$ feet. From Figure 17 the power coefficient is 1.02 for $K_Q=0.22$, and, hence, the required power is $1.02\times 10^{-12}\times 1{,}314^3\times 4.91^5=6.61$ h. p. The desired performance is obtained and the power is only 78 per cent of that required for the first fan, which gave a lower performance. The importance of a proper selection of diameter and speed of rotation is evident.

In many cases this fan would be objectionable on account of the high speed of rotation. To reduce the speed of rotation we would use a fan of higher pitch/diameter ratio, say 0.938, but choose the diameter and speed so as to lie near the condition of maximum efficiency (fig. 19) where $K_H=1.75$, $K_Q=0.24$. Hence

$$D^4=\frac{1.75\times 34{,}200^2}{10^8\times 0.24^2\times 0.5}$$

$$N=\frac{34{,}200}{0.24D^3}$$

from which $D=5.16$ feet, pitch $=4.84$ feet, $N=1{,}035$ r. p. m., power $=6.51$ h. p.

It might be desirable to reduce the speed somewhat more by a sacrifice of efficiency. For example, a convenient motor speed is 900 r. p. m. We would then have the problem of selecting D such that $10^8\frac{0.5}{900^2\,D^2}=K_H$ and $\frac{34{,}200}{900\,D^3}=K_Q$ are points on the characteristic curve of Figure 19. By trial and error, assuming $D=5.3$, 5.4, 5.5, 5.6, 5.7, 5.8, 5.9, 6.0 feet successively, we find $D=5.78$ feet gives $K_H=1.85$, $K_Q=0.197$, $K_P=1.58$ as the solution. The pitch is therefore 5.42 feet and the power required is 7.43 h. p. The additional 0.92 h. p. is the price paid for the reduction of the speed to 900 r. p. m.

In theory, any condition of head and volume may be met efficiently by the use of 2-blade propeller fans; for example, suppose that it is required to select a 2-blade propeller fan to deliver 75,000 cu. ft./min. against a total pressure of 10 inches of water. Again using the fan of pitch/diameter ratio 0.938 for which at maximum efficiency $K_H=1.75$, $K_Q=0.24$, $K_P=1.6$, we have

$$D^4=\frac{1.75\times 75{,}000^2}{10^8\times 0.24^2\times 10}$$

$$N=\frac{75{,}000}{0.24D^3}$$

from which we find readily that a fan of diameter 3.61 feet, pitch 3.39 feet, rotating at 6,640 r. p. m., absorbing 287 h. p., meets the requirements. In practice, the high rotational speed makes the use of a propeller fan impracticable, except, perhaps, under unusual

circumstances, because of the problems of motive power at the desired speed, noise, and strength to resist centrifugal force.

The limits of the practical use of propeller fans may be fixed in the following manner: We may decide that the tip speed should not exceed 25,000 feet per minute (an illustrative figure only, since in most cases lower values are desirable). The maximum value of K_H for good efficiency is of the order of 2.0. Hence, the maximum value of the head under the restriction on the tip speed is $2.0\times10^{-8}\times\left(\frac{25,000}{\pi}\right)^2 = 1.26$ inches of water. In round figures, 2-blade propeller fans are not suitable for pressures much greater than 1 inch of water. The propeller fan is adapted to move large volumes of air against relatively small pressures.

This limitation may be overcome by the use of more than two blades or by the use of a number of fans in tandem. Test data are not available on the performance obtained with these arrangements. The reason for the limitation is apparent. The blades which force the air against the pressure occupy only a small part of the duct area and the air leaks back between the blades. Only by moving the blades very rapidly, indeed, can large pressures be maintained.

VIII. ACCURACY OF RESULTS

Table 4 illustrates the degree of uniformity of the velocity over the measuring cross section, by giving for various conditions the mean percentage deviation of the 40 readings from the mean. It will be noted that the maximum deviation for the blower-fan arrangement is about 17 per cent. Such large deviations would seem to be a fruitful source of error, but when check runs were made it was possible to repeat the results within about 2 per cent. Table 2 (*g*) and Figure 18 show a large number of check observations. In this case, the obstructing netting was changed in amount or was readjusted between tests. Consequently, the resistance offered by it in the nominally identical series may differ significantly, depending, as it does in part, upon the amount of overlapping and of accumulated dirt. Hence, these check observations actually correspond to different points upon the characteristic curve. Tables 2 (*g*), and 3 (*c*), (*d*), and (*g*) contain several check measurements made successively without changing the netting in any way.

For the exhaust arrangement, conditions are greatly improved. The maximum deviation is only 6 per cent. The improvement is due in part to the faired entrance cone, in part to the fact that the motor and its supports are now downstream and in part to the generally steadier flow in the inlet side of a propeller fan.

It is believed that the values given are correct to within 5 per cent. The relative values for the several fans are probably within 2 per cent. The agreement of the coefficients at two speeds of rotation is an evidence of the general precision.

TABLE 4.—*Mean percentage deviation from average velocity at measuring cross section; various resistance conditions*

BLOWER FAN							
5-foot pitch		6-foot pitch		7.5-foot pitch		8.5-foot pitch	
585 r. p. m.	700 r. p. m.	585 r. p. m.	700 r. p. m.	585 r. p. m.	700 r. p. m.	585 r. p. m.	700 r. p. m.
17.2	15.7	14.7	15.6	14.7	16.6	10.1	16.6
6.5	6.9	7.6	6.9	5.5	7.0	9.3	7.0
6.7	8.4	8.1	9.0	8.8	8.9	5.9	8.9
9.6	7.8	9.1	6.4	9.1	13.0	9.0	13.00
5.7	4.8	4.7	5.3	6.4	5.4	5.9	5.4
EXHAUST FAN							
4.3	3.2	4.6	3.1	3.6	3.1	2.6	3.1
3.3	3.8	3.5	3.6	4.0	3.8	4.1	3.8
4.4	4.0	4.5	4.3	4.3	4.0	4.9	4.4
5.9	5.3	6.6	6.8	4.9	4.7	5.2	4.9
3.6	3.9	3.0	3.1	3.3	3.4	3.0	3.0
4.9	5.5	5.1	5.9	5.4	5.4	5.1	6.0

IX. CONCLUSION

The characteristics of 2-blade propeller fans have been measured under operating conditions approximating those encountered in cooling towers. The results are expressed in the form of head and power coefficients plotted against a volume coefficient in such a manner as to facilitate the estimation of the performance of similar fans of any diameter and speed of rotation. The effect of pitch/diameter ratio on the maximum efficiency obtainable is shown to be small for pitch/diameter ratios between 0.375 and 0.938. The speed of rotation and fan diameter must be carefully chosen to obtain maximum efficiency. Methods of selection are illustrated by numerical examples.

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